HYDRAULIC TURBINE

WATER DEPRESSION SYSTEMS

FOR

SYNCHRONOUS CONDENSER OPERATION

BY

J. R. TAYLOR

THERMAL GENERATION DEPT.

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HYDRAULIC TURBINE WATER DEPRESSION SYSTEM FOR SYNCHRONOUS CONDENSER OPERATIONS

Author: J.R. Taylor - Hydro Electric Power Commission of Ontario

Summary

This paper describes the method and procedure developed by Ontario Hydro for determining the size of the component parts of a compressed air system designed specifically for tailwater depression in hydro-electric plants. Test data is included which shows the effects of design variables on the transient electric power conditions during the period when the generator is being converted to a synchronous condenser.

Introduction

Many of the more recent hydraulic power developments in the Ontario Hydro systems are characterized by three important features. The first is that the plants are designed to operate as peaking plants. Second, they are remotely situated, requiring long transmission lines to carry power to load centres. The third feature is that the total head available at these sites is generally less than 100 feet.

During periods when there is insufficient water to permit operation of the units as generators, the charging currents on the long transmission lines, operating at extremely light loadings, could result in abnormally high voltages. This problem is overcome by operating the generators as underexcited synchronous condensers to absorb reactive power from the system. As the electrical load

on the line is increased, the units may be required to supply reactive power.

When hydraulic heads are as low as previously indicated, turbines with fixed-blade, propeller-type runners are usually employed. To minimize cavitation, the runners must be set below tailwater level.

If the runners are allowed to remain immersed in water when the unit is operating as a motor, loads as great as 30 megawatts can be imposed. This is remedied by using a water depression system to lower the water level in the draft tube and free the turbine runner of all possible load other than friction.

A turbine water depression system may be defined as a system for compressing, storing and controlling a supply of air for displacing the water surrounding the turbine runner, which leaves the runner free to rotate in a pocket of air. Other names applied to such a system include draft tube evacuation, draft tube water level depressor and tailwater depression.

Although water depression systems were first investigated by Ontario Hydro in 1943, it was not until 1956 that the first system was installed in its Manitou Generating Station on the English River. Since that time, water depression systems have been installed at five other stations: Whitedog Falls, Silver Falls, Red Rock Falls, Otter Rapids and Little Long. All these plants are unattended and operated by remote control. The purpose of this paper is to describe the design criteria evolved over this period.

Basically, the water depression system consists of air compressors, air receivers, control air valves, and draft tube water

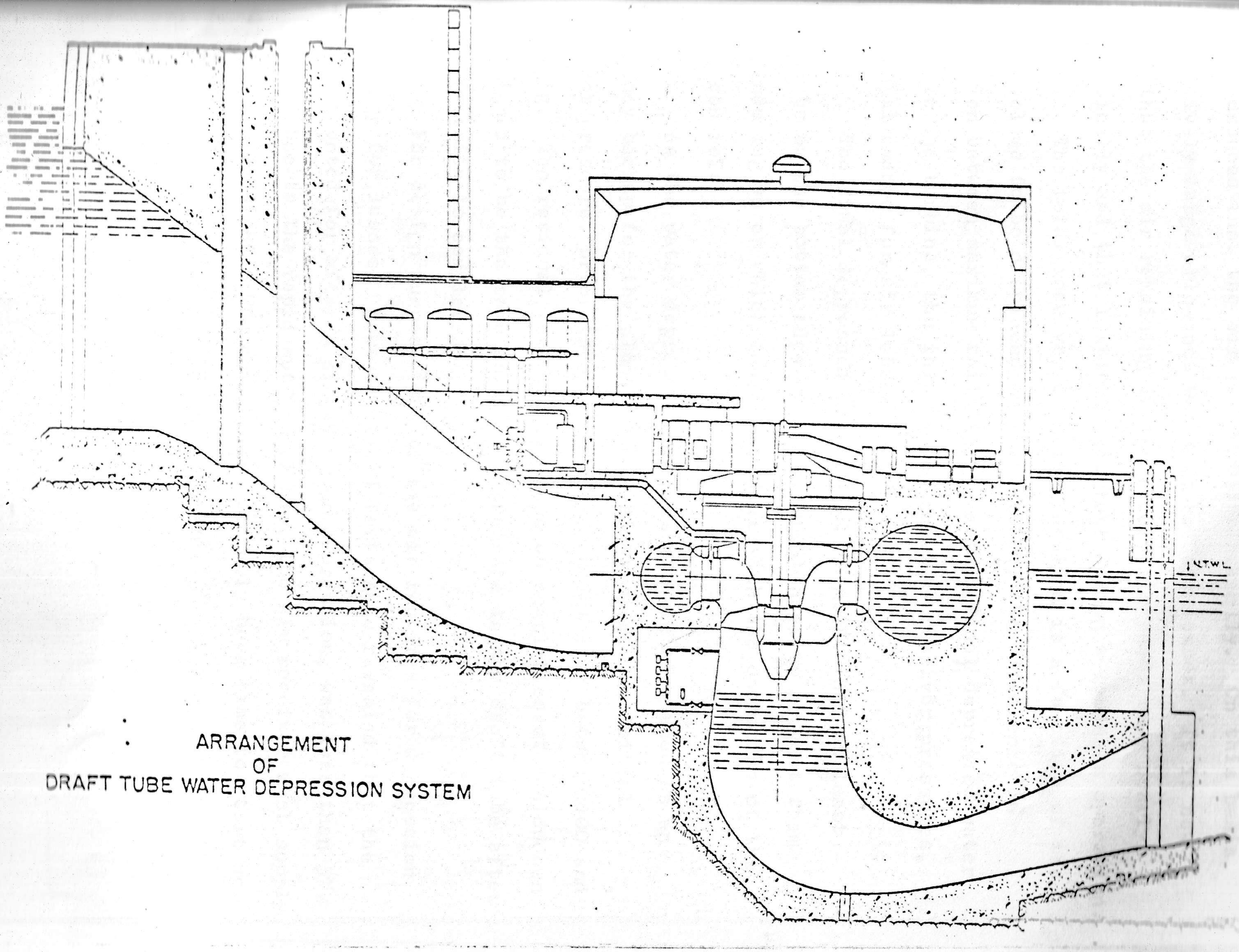
level controls (See Fig. No. 1). To condense* a unit, the receivers are brought up to pressure and the main control air valve is opened. When sufficient air has been admitted to depress the water in the draft tube to a predetermined level, the main control valve is closed. A small make-up valve operates as required to maintain the water level during the condense period.

Two basic methods are used to condense a unit. One is to back off the load on the generator until the wicket gates are fully closed and the unit is motoring in water. The air valves are then opened until the runner is rotating in a pocket of air. With fixed-blade propeller runners, this method requires that the unit be supplied with a large amount of backfeed power until depression has been completed.

The other method, used by Ontario Hydro, is to back off the load on the generator until the turbine wicket gates are at the synchronous no load speed position. The air valves are opened and after a short time delay, the wicket gates automatically close. When properly applied, this method results in a relatively smooth transition from generating to condensing as far as power surges are concerned. However, it has been found that this latter method requires a more rapid admission of air to minimize power surge.

Although the first method has a lower initial cost, due to smaller piping requirements, the second method results in lower power losses each time the unit is condensed. The savings in power costs will soon recover the difference in the initial cost. Also harmful vibration caused by shaft-lift planing action is reduced.

* The term "condense" refers to the procedure of converting a generator to a synchronous condenser.



ARRANGEMENT
OF
DRAFT TUBE WATER DEPRESSION SYSTEM

6

FIG. N

Air Receivers

In our first attempt to determine air receiver requirements, it was assumed that the air expanded isothermally. On this basis, a relatively large allowance was made for air losses. It is now known that the air remaining in the receiver expands very nearly adiabatically and that little air is lost during a condense operation.

The total receiver capacity required is a function of the water volume to be displaced in the turbine, the difference in elevation between maximum tailwater level and final depressed water level in the turbine, the initial and residual receiver pressures, and an allowance for air losses.

The final depressed water level is maintained at least 5 feet below the bottom edge of the runner blades to prevent them from inducing waves which could interfere with the operation of the level controls.

As the water depression compressors are also used as an emergency backup for the plant service air system, the initial receiver pressure, prior to a blowdown, is maintained between 100 and 110 psig. The residual receiver pressure on completion of a blowdown is designed to be the same as the air pressure in the draft tube after the depression is completed.

For design purposes 10% of the air required for depression is allowed for losses. However, field tests have revealed that the air unaccounted for after a single water depression, was less than 5%.

Due to the rapid outflow of air, the air receiver must supply all the air required. The compressors scarcely have time to start up and load before the depression is completed.

Space limitations in the powerhouse usually dictate the size and number of receivers required to contain a predetermined volume of air. These receivers (sometimes four or more) are inter-connected, without shut-off valves, in such a way as to represent a single, large receiver. It has been standard practice in the past to locate the receivers indoors but in two stations now under construction they will be located outdoors.

Air Compressors

The compressor capacity is determined by the quantity of air used for one depression, the volume of air required to hold a unit on the line as a condenser and the minimum time interval between placing any two units into similar service. When a hydro-electric station contains two or more generators, an interval ranging from 30 to 45 minutes is usually required between the placing of any two units on the line as condensers. The air required to hold a unit on condense has been found to vary from 40 to 100 standard cubic feet per minute. It is believed that the major source of air loss is a result of the air being soluble in water. The water leaking through the closed wicket gates is sprayed into a chamber of compressed air and is readily aerated. The water entering the turbines in most plants is taken in many feet below the surface of the forebay where, particularly in winter, the water is relatively free of dissolved gases. Air loss is mainly dependent on the amount of gate leakage, water temperature, and is proportional to the air pressure in the turbine passages.

Control Valves

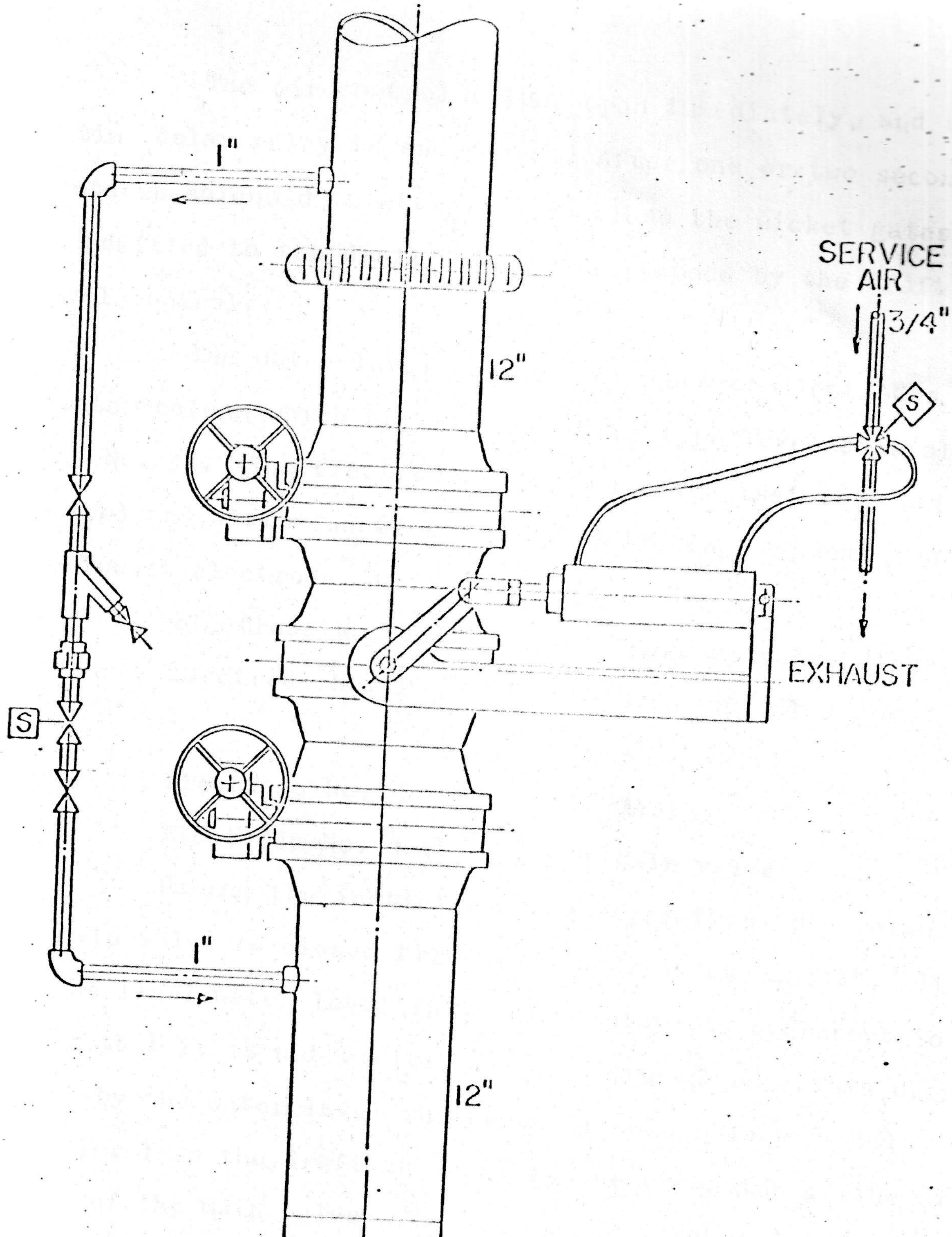
The air control valves for each unit consist of one main valve and one small make-up valve (See Fig. No. 2). The main valve is used during the initial water depression and also serves as a backup to the make-up valve. The function of the make-up valve is to maintain the water level below the runner for the duration of the condense period.

Initially, diaphragm-operated control valves were used for the main valves, but these were later discarded in favour of pneumatic cylinder-operated butterfly valves with rubber seats. The cylinder operators provide rapid opening and the butterfly type valve offers the advantage of a low pressure drop. The pneumatic cylinder is operated by a four-way solenoid valve which is connected to the station's service air system. This is done to ensure adequate pressure for closing the valves. For ease of maintenance, the main valve is located between manual isolating valves of the gear-operated butterfly type. Except during maintenance the isolating valves remain open.

The make-up valve consists of a direct operated solenoid valve installed in a by-pass around the main valve.

Control

Interlocks are provided to ensure that the circuit breaker for the unit is closed, the turbine wicket gates are at the synchronous no load speed position, and the air pressure in the receivers is adequate before the unit can be operated as a synchronous condenser. To initiate a condense operation, the unit is brought to the synchronous no load speed position and the condense control switch is moved to the ON position.



PICAL WATER DEPRESSION AIR CONTROL VALVES

The air control valves open immediately, and simultaneousl a time delay relay is energized. After one or two seconds, the unit shutdown solenoid is energized to close the wicket gates. The air is admitted to the draft tube until stopped by the draft tube water level control.

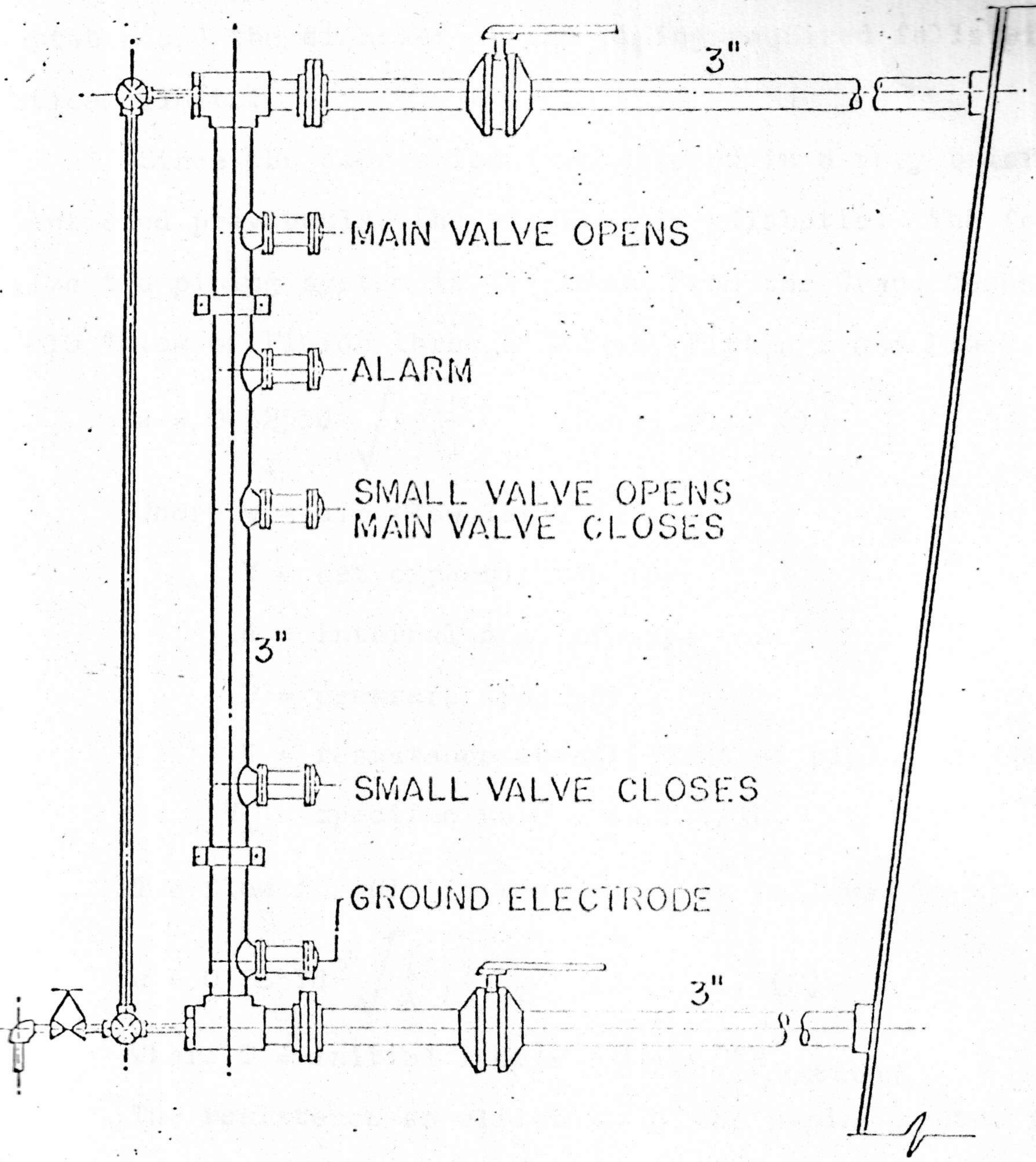
The water level control consists of electrodes located in a water column which operate inductive type level control relays (See Fig. No. 3). The electrodes are so located that they actuate the control and alarm functions in the following sequence, starting with the lowest electrode:

Electrode No. 1 (Lower)	Make-up valve closes.
Electrode No. 2	Make-up valve opens, main valve closes.
Electrode No. 3	Alarm.
Electrode No. 4 (Upper)	Main valve opens.

During the initial condense operation both valves are open. The main valve is closed first to minimize "overshoot". It is possibl for the final water level in the draft tube to overshoot to such an extent that it is not visible in the gauge glass. This overshoot is caused by the water level in the water column lagging behind the water level in the draft tube and by the expansion of the air downstream of the main valve after the valve is closed.

Sizing of Piping System

The rate of air discharge from the receivers has been found to have a definite effect on the power requirements of the generator during the period when air is being admitted to the draft tube. A high rate of air discharge provides a smooth transition from generat to condense as far as power conditions are concerned.



WATER DEPRESSION CONTROL COLUMN

FIG. N°. 3

(13)

The rate of air discharge is designed so that the water level is depressed below the runner blades in 10 seconds or less. On this basis, the resulting transient power inputs are considered acceptable and the diameter of the piping required falls within practical limits.

Since the depression is completed in a very brief period, as mentioned previously, the air flow is adiabatic. The formula used to size the piping system is developed from the Crane Technical Paper No. 410 "Flow of Fluids through Valves, Fittings and Pipe".

$$w = 0.525Yd^2 \sqrt{\frac{\Delta P}{KV_1}} \quad \text{.........} \quad (1)$$

Where w = air flow lbs./sec.

Y = net expansion factor

d = internal dia. of pipe ins.

P = pressure drop psi

K = resistance co-efficient of piping system

V_1 = specific volume cu.ft./lb.

The flow formula is rearranged as follows:

$$w = 0.863Yd^2 \sqrt{\frac{(\Delta P)}{(\ P)} \frac{P^2}{KT}} \quad \text{.........} \quad (2)$$

Where T = initial receiver temp. °R ABSOLUTE

The resistance co-efficient of the piping system is determined from:

$$K = \frac{12\ fL}{d} \quad \text{.........} \quad (3)$$

Where f = friction factor

L = equivalent length of piping system in feet.

Due to the receiver pressure decreasing during the blow-down period, a simplified method is used to check the time required to clear the runner blades. In the above formula the Y and $\Delta P/P$ terms are considered to be constant for any value of K (See Fig. No. 4). The values used are those for which sonic velocity occurs at the discharge end of the pipe. The effect of the increasing draft tube pressure on the flow rate is relatively small and is not included in the calculations.

For an assumed piping diameter then, the flow formula is reduced to:

$$W = C_1 P \qquad \text{.........} \quad (4)$$

The air supplied from the receivers is calculated from the following formula:

$$W = \frac{144 P_1 V_1}{R T_1} \left[1 - \left(\frac{P_2}{P_1} \right)^{.717} \right] \qquad \text{.........} \quad (5)$$

Where W = lbs. of air

P_1 = initial receiver pressure

P_2 = final receiver pressure

R = gas constant (53.3 for air)

T_1 = initial receiver temperature (°F + 460)

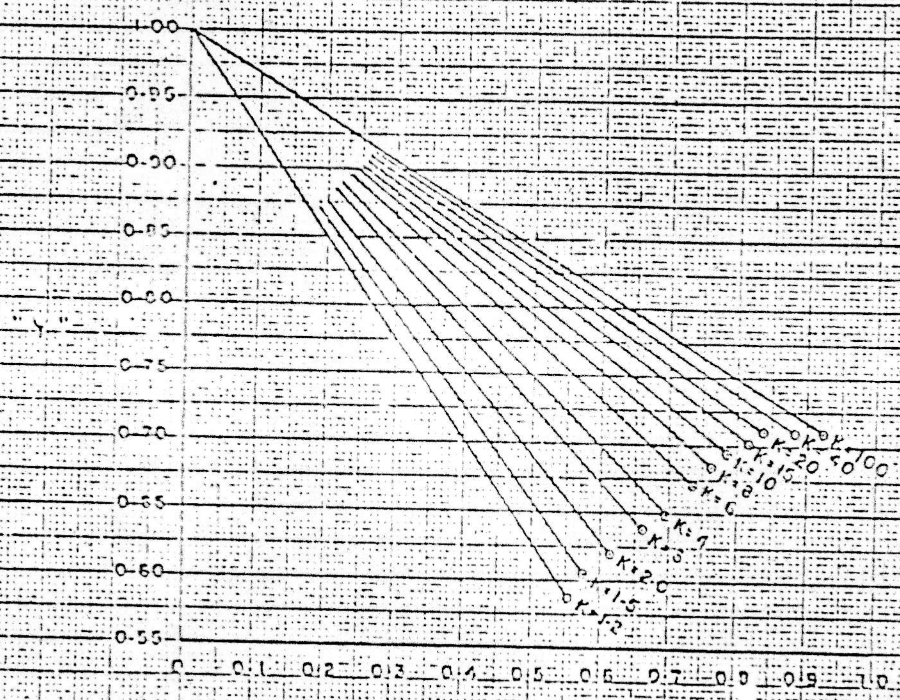
Equation (5) was used to plot the curve shown in Fig. No. 5. This curve is based on initial receiver pressure and temperature conditions of 100 psig and 70°F and for a receiver volume of 1,000 cubic feet. From this curve it is evident that the weight of air available can be closely represented by:

$$W = C_2 (dP) \qquad \text{.........} \quad (6)$$

LIMITING FACTORS FOR SONIC VELOCITY

$k = 1.4$

"K"	$\frac{\Delta P}{P'}$	"Y"
1.2	552	588
1.5	576	606
2.0	612	622
3	662	639
4	697	649
6	737	671
8	762	685
10	784	695
15	818	702
20	839	710
40	883	710
100	923	710

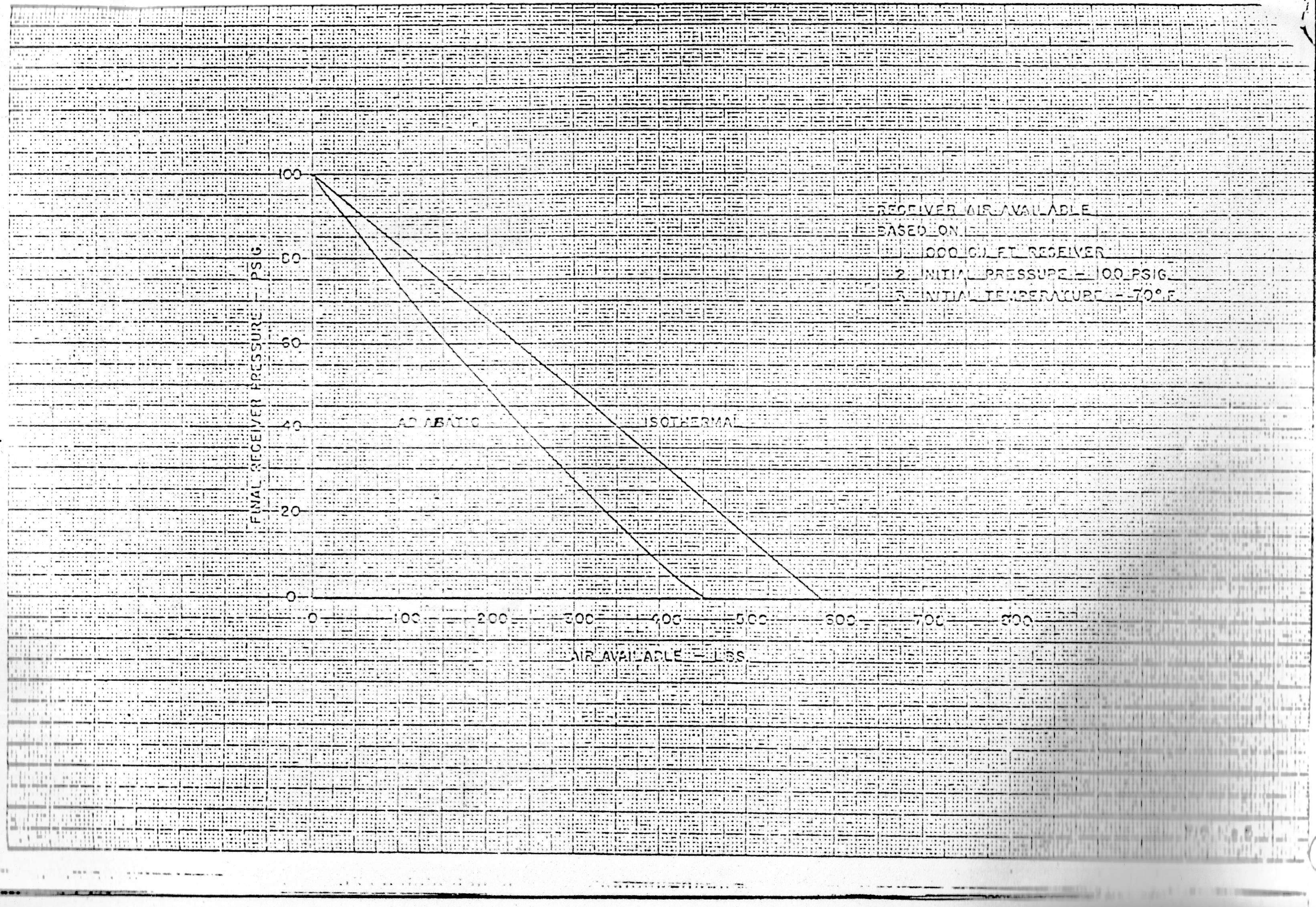


$\frac{\Delta P}{P'}$

FROM CRANE TECHNICAL PAPER No. 410

NET EXPANSION FACTOR "Y"
FOR COMPRESSIBLE FLOW
(k = 1.4 ADIABATIC)

FIG. No. 4



RECEIVER AIR AVAILABLE
BASED ON
1000 CU FT RECEIVER
2 INITIAL PRESSURE – 100 PSIG
3 INITIAL TEMPERATURE – 70°F
FINAL RECEIVER PRESSURE – PSIG
100
80
60
40
20
0
ADIABATIC
ISOTHERMAL
0
100
200
300
400
500
600
700
800
AIR AVAILABLE – LBS

 (17)

Combining equations (4) and (6) in differential form

$$C_1 P(dt) = C_2(dP) \qquad \text{.........} \quad (7)$$

Rearranging and integrating between the initial and final eceiver pressures

$$t = \frac{C_2}{C_1} \log_e \frac{P_1}{P_2} \qquad \text{.........} \quad (8)$$

Where t = time in seconds

Although equation (8) is lacking in several theoretical spects, it has been found to give close approximation to the values btained from field tests.

See the appendix for a sample calculation of time required o depress the water level to the underside of the runner blades.

ield Tests

A number of tests were conducted at our Little Long enerating Station, located on the Mattagami River approximately 45 iles north of Kapuskasing, which was placed in service in the fall f 1963. The main purpose was to determine the shape of the receiver ressure "decay" curve and to measure the transient power conditions uring a condense operation. A test was also made to verify that the ir expansion in the receivers during a blowdown was adiabatic.

A Texas Instruments Corporation portable, two-channel scillo-Riter" recorder was used to record the shape of the receiver essure decay curve and to obtain the transient power conditions. e transient power conditions were recorded for different intervals ween the time the air valves opened and the wicket gates began sing. In addition, the effect of a slower air admission rate on r conditions was obtained by throttling the air valves.

A conventional resistance bridge-type pressure transducer was installed on the air receiver and was connected to the recorder by shielded cable. Fig. No. 6 shows a typical receiver pressure decay curve and for comparison the theoretical curve has been plotted. The theoretical curve does not allow for the opening time required for the main valve and therefore the test curve lags behind the theoretical curve.

The transient power conditions were sensed by a Bristol Thermoverter. This device was connected to the instrument transformers on the generator ac metering system and provided a dc output proportional to the power in the ac system.

The effect of different time delay settings on the transient power conditions is shown in Fig. No. 7. The longer time delay causes the unit to momentarily generate up to 8 megawatts. The generation of power is attributed to the additional head imparted to the water flow through the turbine by the air admission.

The effect of throttling the air supply on the transient power conditions is shown in Fig. No. 8. Test curve No. 11 shows a comparatively smooth transition. The turbine blades for this test were calculated to be clear of water in approximately 5-1/2 seconds. Test curve No. 17 indicates a much rougher transition with the blades cleared of water in a calculated time of 11-1/2 seconds. For design purposes we have adopted a blade clearance time of not more than 10 seconds.

Both time delay and throttling tests appeared to have no significant effect on the amount of air required to depress the water level or, conversely, the amount of air lost or otherwise unaccounted for.

The test to prove adiabatic expansion of air in the receiver consisted of bringing the receiver up to pressure, pulling the disconnects on the compressor motors and condensing a unit. The air pressure in the receiver was recorded (See Fig. No. 9).

The initial receiver conditions were 103.2 psig and 73°F. Immediately after the main air valve closed, the receiver pressure was 29.3 psig. After 50 minutes the receiver pressure increased to 42.1 psig as the air remaining warmed up to ambient temperature.

From the general law PV^N = a constant, the exponent N was calculated from this test to be 1.35. This compares with an exponent of 1.4 for the true adiabatic. It is believed that if the piping system had been completely free from leaks, the final air pressure after warm-up would be higher and the value of N very nearly equal to 1.4. The estimated air temperature in the receivers after blowdown is -57°F based on adiabatic expansion.

Conclusions

1) The calculated pressure decay curve is close enough to the test curve for all practical purposes.

2) The tests indicated that the optimum time delay setting was approximately 1.5 seconds.

3) To obtain minimum transient power variation, the rate of air discharge should be as rapid as possible.

4) The expansion of air in the receiver occurs very nearly adiabatically.

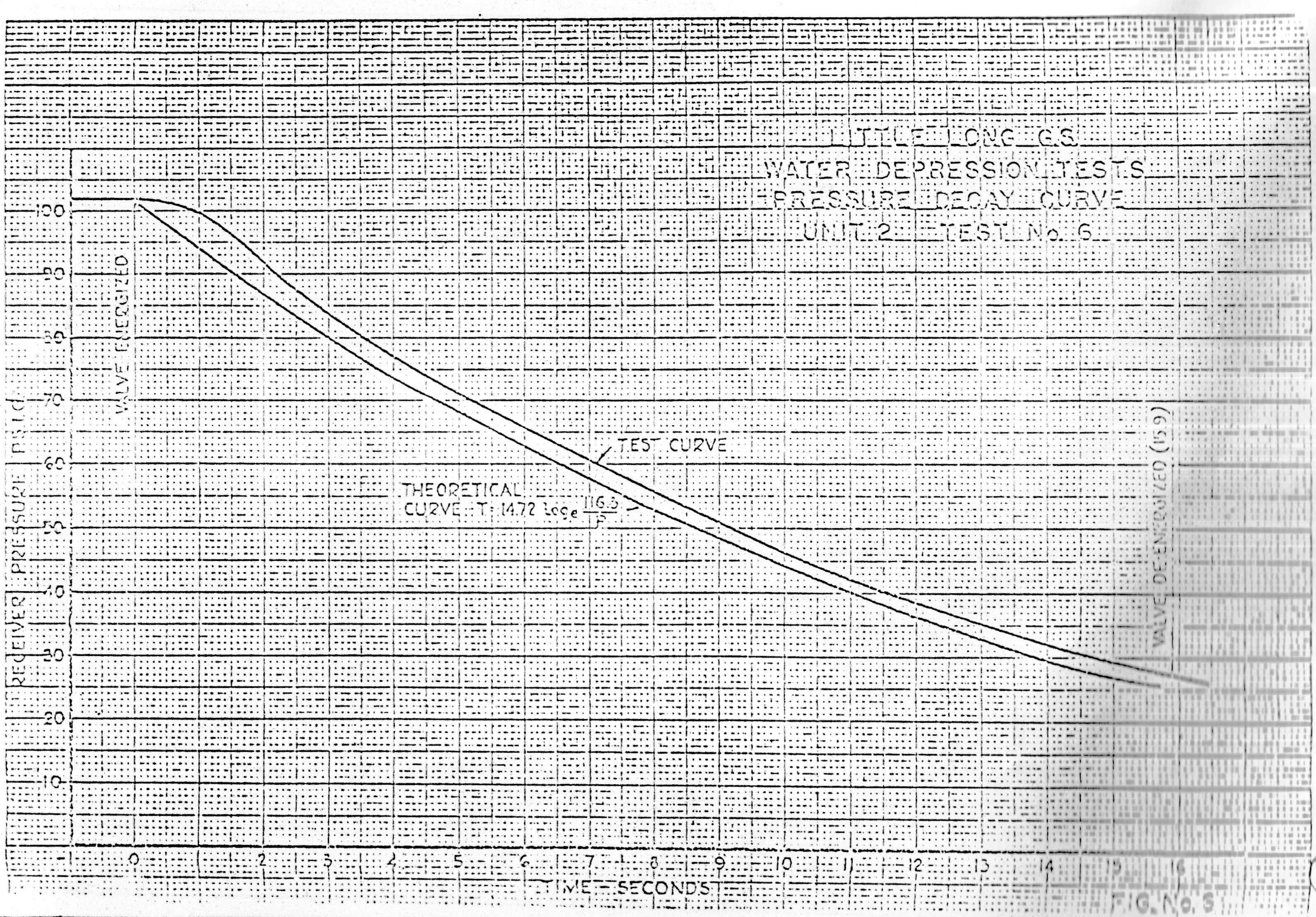

LITTLE LONG G.S.
WATER DEPRESSION TESTS
PRESSURE DECAY CURVE
UNIT 2 TEST No. 6
VALVE ENERGIZED
VALVE DE-ENERGIZED (15.9)
TEST CURVE
THEORETICAL CURVE T = 14.72 $\log_e \frac{116.5}{P}$
RECEIVER PRESSURE P.S.I.G.
100
90
80
70
60
50
40
30
20
10
-1
0
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
TIME – SECONDS

FIG. No. 5

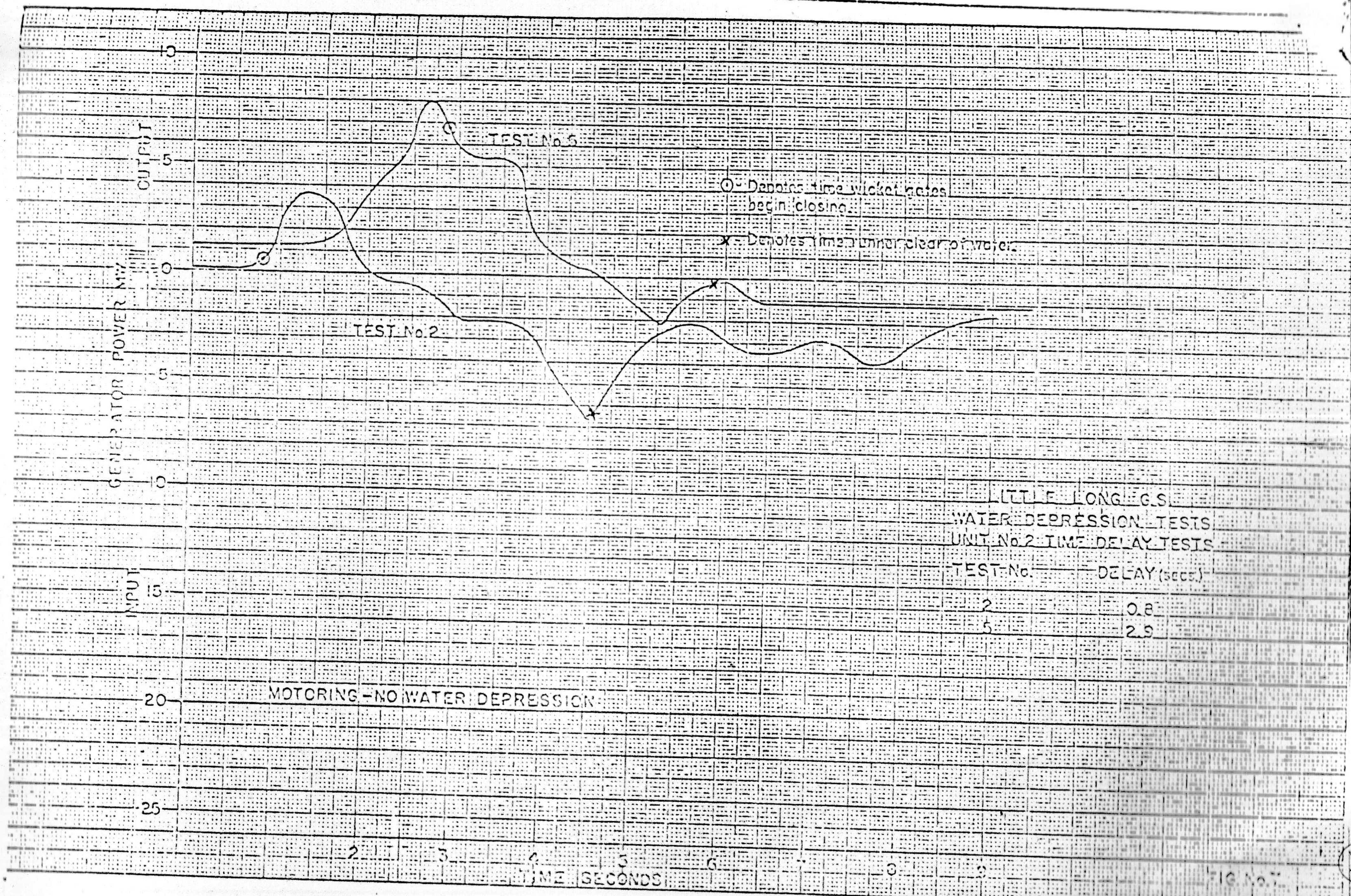

LITTLE LONG G.S.
WATER DEPRESSION TESTS
UNIT No.2 TIME DELAY TESTS
TEST No. DELAY (secs.)
2 0.8
5 2.9
⊙ - Denotes time wicket gates begin closing.
x - Denotes time runner clear of water.
TEST No.5
TEST No.2
MOTORING - NO WATER DEPRESSION
GENERATOR POWER MW
OUTPUT
INPUT
10
5
0
5
10
15
20
25
2
3
4
5
6
7
8
9
TIME SECONDS

FIG No. 7

21

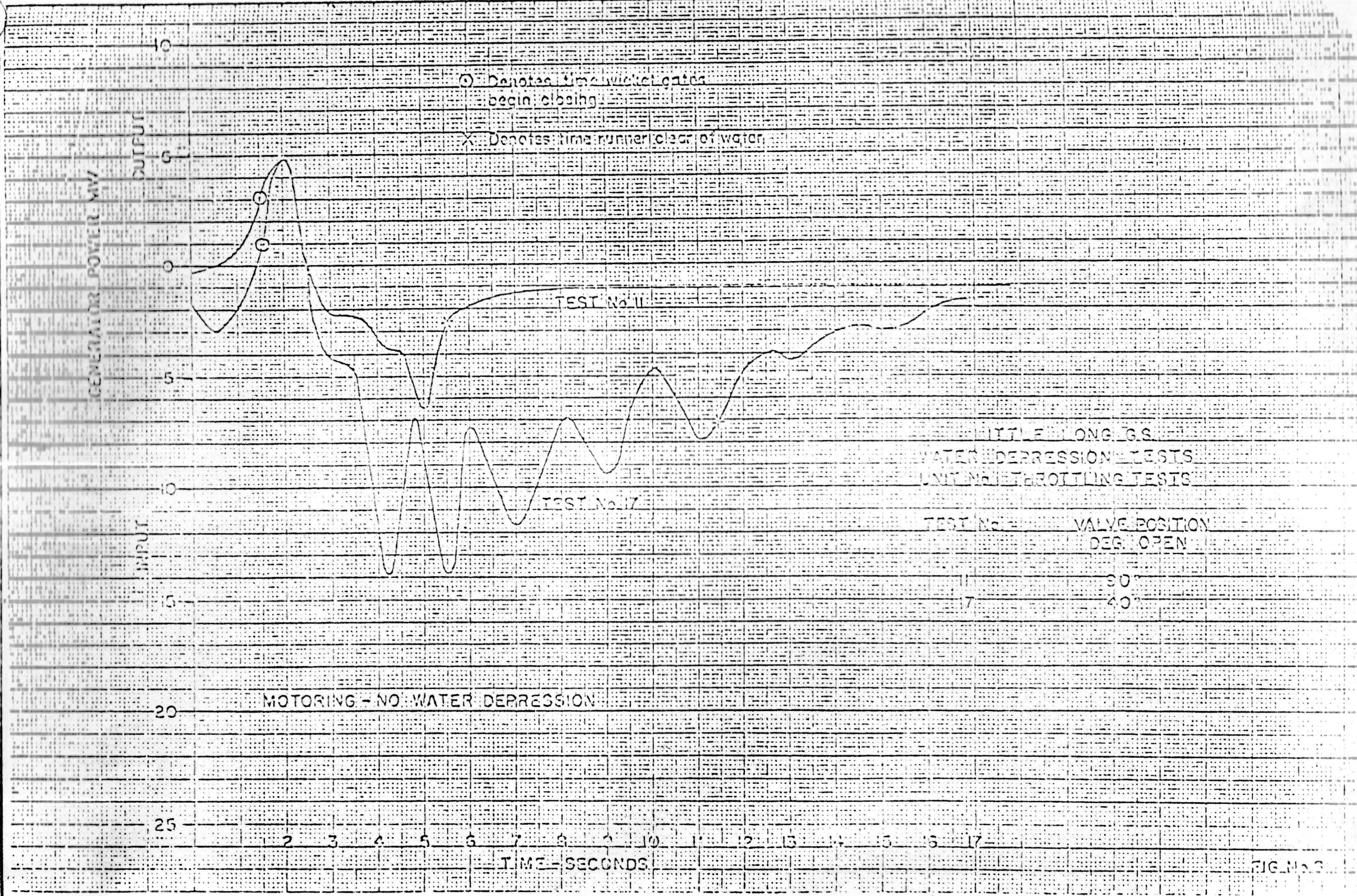

GENERATOR POWER MW
OUTPUT
INPUT
10
5
0
5
10
15
20
25
⊙ Denotes time wicket gates begin closing
X Denotes time runner clear of water
TEST No.11
TEST No.17
MOTORING - NO WATER DEPRESSION
2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17
TIME - SECONDS
ITTLE LONG G.S.
ATER DEPRESSION TESTS
THROTTLING TESTS
TEST No.
VALVE POSITION
DEG. OPEN
11
17
30°
40°

FIG. No. 3

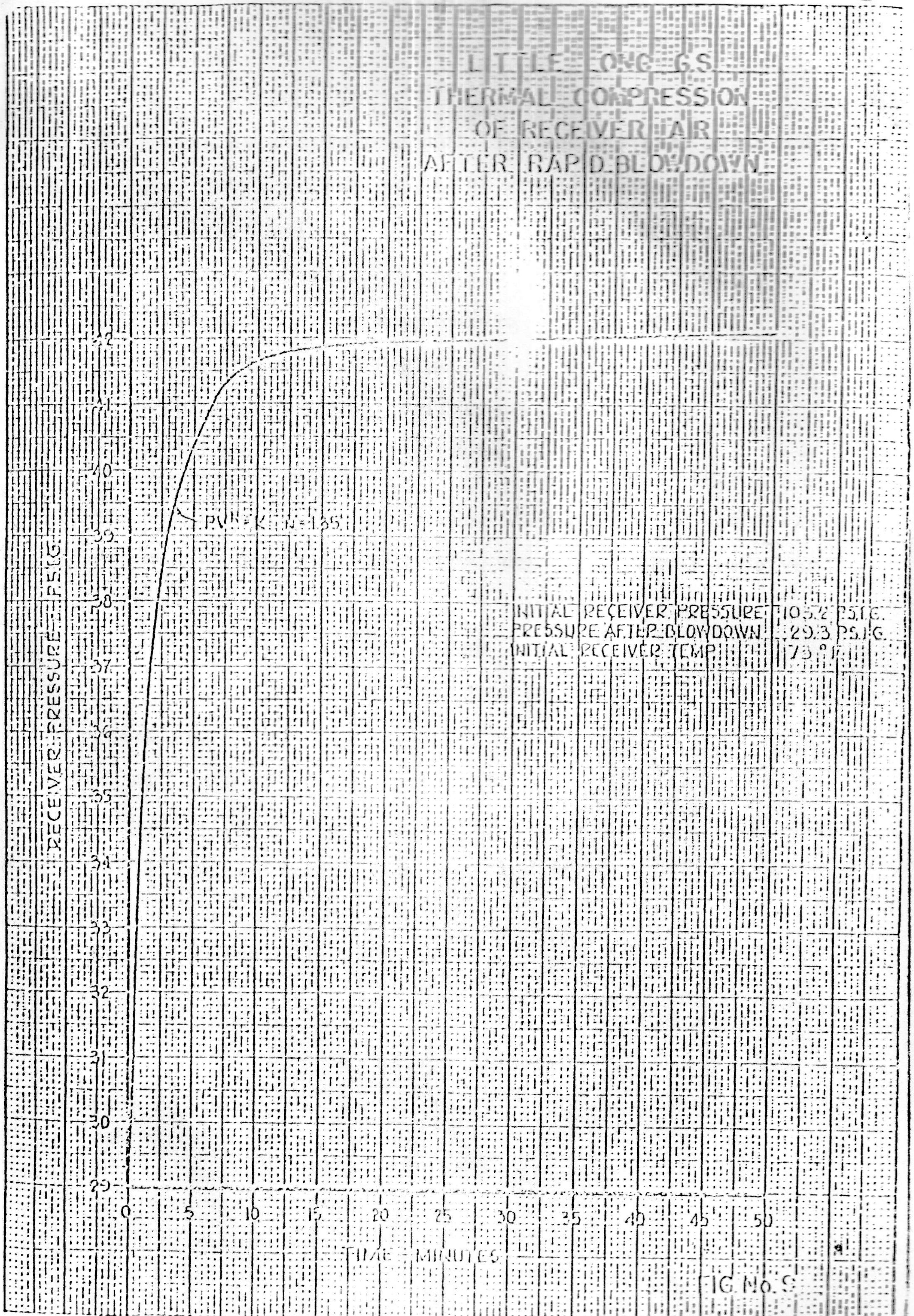

LITTLE LONG G.S.
THERMAL COMPRESSION
OF RECEIVER AIR
AFTER RAPID BLOWDOWN
RECEIVER PRESSURE - PSIG
TIME - MINUTES
$PV^n = K \quad n = 1.35$
INITIAL RECEIVER PRESSURE 105.2 PSIG.
PRESSURE AFTER BLOWDOWN 29.3 PSIG.
INITIAL RECEIVER TEMP 73 °F
29
30
31
32
33
34
35
36
37
38
39
40
41
42
0
5
10
15
20
25
30
35
40
45
50

FIG No. 9

(24)

ENDIX

Following is a sample calculation of the time required to press the water level to the underside of the runner blades.

Assume – Receiver volume	– 4000 cu.ft.
Initial receiver pressure	– 100 psig
Initial receiver temperature	– 70 F
Inside diameter of pipe	– 12 ins.
Piping resistance coefficient (K)	– 8.0
Weight of air in draft tube in and above runner	– 600 lbs.

The weight of air required per 1000 cu.ft. of receiver olume is 600 ÷ 4 or 150 lbs. From Fig. No. 5 the final receiver pressure is 61.0 psig.

Using equation (6) $W = C_2\, dP$

$$600 = C_2\,(100-61.0)$$

Therefore $C_2 = 15.4$

or $W = 15.4\,(dP)$

From Fig. No. 4 for a piping system resistance coefficient of 8.0, the limiting factors for sonic velocity are .762 for $\Delta P/P$ and .685 for Y.

From equation (2) $W = .863Yd^2\sqrt{\frac{\Delta P}{P}\frac{P^2}{KT}}$

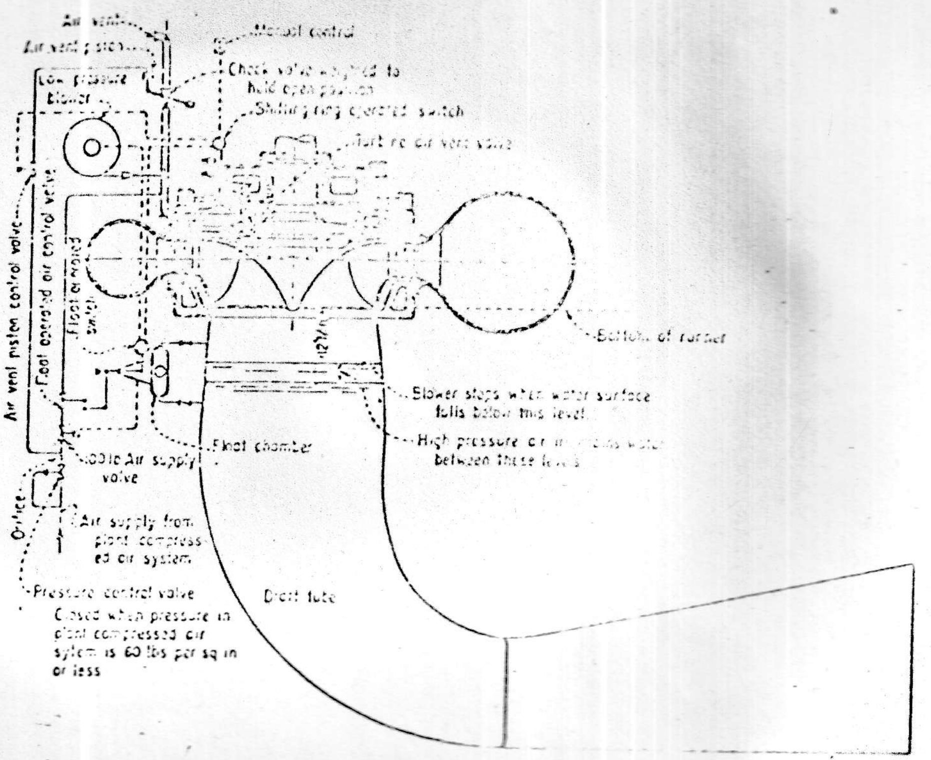
$$W = .863 \times .685 \times 12^2\sqrt{\frac{.762\ P^2}{8.0 \times 530}}$$

or $W = 1.14P$ (equation 4)

From equation (8) $t = \frac{15.4}{1.14}\log_e\left(\frac{100 + 14.7}{61.0 + 14.7}\right)$

$$t = 5.6 \text{ seconds}$$

HYDRAULIC TURBINES--DRAFT-TUBE WATER-LEVEL DEPRESSOR



Provision shall be made for a depressor system if normal tailwater of a plant is above the bottom of the runner. The operation should be fully automatic, with a manual control permitting the operator to cut out the depressor when desired. A low pressure blower is preferred for initial underwatering, and can also be used to pressurize the turbine air vent if found necessary by field tests (See X-D-3310). Large air storage tanks may be used instead of the blower if station service power supply is insufficient to carry peak load of blower motor.

Design Conditions

1. Underwater runner in 30 seconds, against a maximum static head of [illegible] above ℄ of float chamber by low pressure air from blower. See Note 1 and Note 3.
2. Maintain depressed level while motoring unit, by controlling inflow of air from plant compressed air system. A [illegible] inch air line is sufficient to replace leakage. See Note 2.
3. Introduce air thru turbine air vent line. Turbine air vent valve is normally in open position when gates are closed. Do not provide auxiliary ports in draft tube inlet to introduce air.
4. Closure of turbine gates automatically starts operation of depressor system. Opening of turbine gates automatically stops operation of depressor system.
5. A manual cutout is required to prevent operation if tailwater is below bottom of runner or above normal.

Operation – The water level in draft tube is depressed by introducing air under pressure thru the turbine air vent piping. The operation is as follows:

1. Closure of turbine wicket gates will
   (a) Close turbine air vent check valve by an auxiliary piston or motor
   (b) Start blower
2. When runner is unwatered, the float control
   (a) Stops the blower,
   (b) Starts flow of air from plant compressed air system
   (c) Maintains depressed water level
3. Opening of turbine wicket gates
   (a) Opens turbine air vent check valve
   (b) Rising float stops flow of air from plant compressed air system
   (c) Opens blower power circuit

Notes

1. If air vent pressurizing is indicated, sufficient blower capacity should be included to meet design condition 3 of X-D-3310.
2. An orifice and pressure control valve in the air supply line from the plant is desirable to limit demand to capacity of the small compressor.
3. Provision shall be made to insure a minimum running time of one minute for proper lubrication of the blower.

HYDRAULIC TURBINES
DRAFT-TUBE WATER-LEVEL DEPRESSOR

delegación
manda a Of. Técnica
de Tecno...

9

COMIPAL
COMISION MIXTA DEL PALMAR

Montevideo, 9 de noviembre de 1982

Sr. Gerente de Sector de la División
Generación y Trasmisión de U.T.E.
Ing. Jorge Fontana
P r e s e n t e

SIRVASE CITAR
NOTA Nº 637/82

OV/mv

De mi consideración:

Para su conocimiento y demás efectos remito a usted copia del artículo "Hidraulic Turbine Water Depression Systems for Synchronous Condenser Operation" de la "Hidro-Electric Power Commission of Ontario".

Sin otro particular, saludo a usted muy atentamente.

COMIPAL
REG[illegible]VO
9 NOV. 1982
Asunto Nº 811/82

~~Ing. Siegmund Antmann~~
Director de Oficina Montevideo

U.T.E.
RECIBIDO
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